10th International Conference
12-14 July 2010
Southampton
RASD 2010

NUMERICAL EVALUATION OF THE INSERTION LOSS OF LIGHTWEIGHT TORHEX SANDWICH PANELS

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Keywords: lightweight panels, vibro-acoustic analysis, insertion loss, TorHex, numerical simulation

ABSTRACT

This paper documents the vibro-acoustic properties of a sandwich panel with a TorHex core and skins of polypropylene, reinforced with natural fibers. It is intended to be in line with the actual NVH (Noise Vibration and Harshness) requirements of lightweight material components, always more crucial in the automotive industry. The IL (insertion loss) of the mentioned structure is experimentally and numerically determined in a frequency range up to 1,4kHz. The experimental analysis is performed by exciting a rigid acoustic cavity enclosed by the sandwich panel and measuring the radiated power. The physical behaviour of the examined structure is well captured numerically in the studied frequency region. Further optimization of the numerical model is on-going in order to get a powerful tool for the prediction of the insulation properties of such a complex structure.

1. INTRODUCTION

The increasing interest towards the application of lightweight structures in transportation vehicles pushed the authors towards the research shown in this paper. Lightweight structures are able to offer important weight saving. Unfortunately they often have bad NVH performance. In view of the "analysis leads design" trend the characterization [1, 2], of such lightweight structural components in terms of radiation and transmission is needed in the first design phases. Detailed investigations are required at the first design stages in order to avoid time and cost expensive corrections at later stages, which could completely cancel out the initial weight reduction. Sandwich structures are already quite wide spread in the automotive industry [3–5]. By adjusting the material and geometric parameters of different layers and cores, it is possible to design a satisfying product for several applications [6]. The research presented in this paper focuses on the evaluation and prediction of the vibro-acoustic characteristics of a TorHex sandwich panel, which consists of a corrugated cardboard core, produced with an automated, fast and continous in-line production process [7–9], covered by layers of polypropylene reinforced with natural fibers. Its transmission properties are experimentally determined and on this basis a valid numerical model is developed in order to get an efficient and fast tool for the prediction of the vibro-acoustic performance of such a complex structure. This study is particularly focused on the prediction of the IL (Insertion Loss), defined by the ratio of the radiated power without enclosure of the sound source, $P_{without\ panel}$, to the power radiated by the same system when the source is enclosed by the structure of interest, $P_{with\ panel}$.

$$\mathbf{IL} = 10 \log \frac{P_{without\ panel}}{P_{with\ panel}} \tag{1}$$

The first part of this work describes the experimental set-up and its results. The second part focuses on the numerical FE (Finite Element) and BE (Boundary Element) models to determine the radiated power. The third section shows the comparison between the experimental and the numerical results. Future studies will focus on the improvement of the numerical model and its applications for sensitivity analyses, in order to optimize the ”core/skin layer” design.

2. EXPERIMENTAL STUDY OF INSERTION LOSS

The examined sandwich panel consists of a TorHex paper honeycomb core and faces reinforced with polypropylene natural fibres. Its geometrical features are shown in Figure 1 and listed in Table 1.

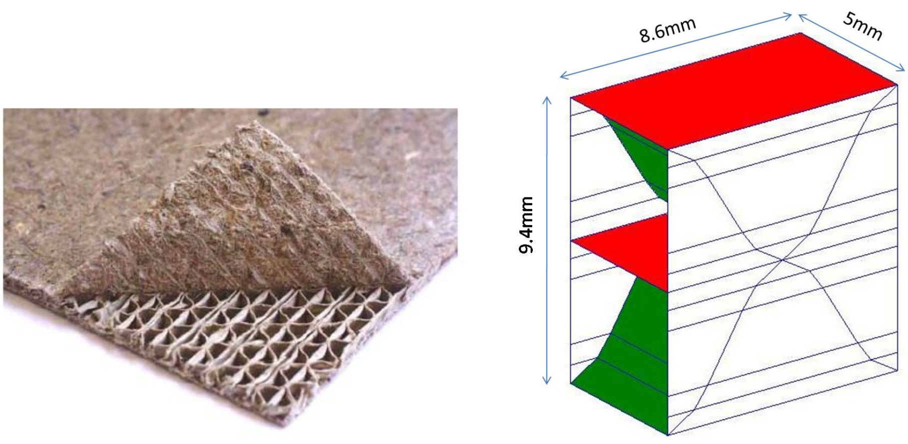


Figure 1: On the left a sample of a TorHex sandwich panel with faces of PP/natural fibers – On the right some geometrical details of the unity cell

length	496mm
width	387mm
thickness	5mm

Table 1. TorHex sandwich panel geometrical sizes

2.1 Test set-up

The test set-up, realized in a semi-anechoic laboratory setting, consists of a box, representing a rigid acoustic cavity, with a built-in loudspeaker. The box comprises two acoustic cavities. The upper cavity has a size of 400mm x 300mm, while the height is 315mm for the largest part. The lower cavity has no other purpose than hosting the loudspeaker, as shown in Figure 2. Standing waves in the cavity are strongly linked to the box dimensions (i.e. the upper cavity) listed above. The acoustic modes will influence the acoustic IL, however, also in reality the applied panels will be coupled to closed cavities and hence, the developed setup is representative.

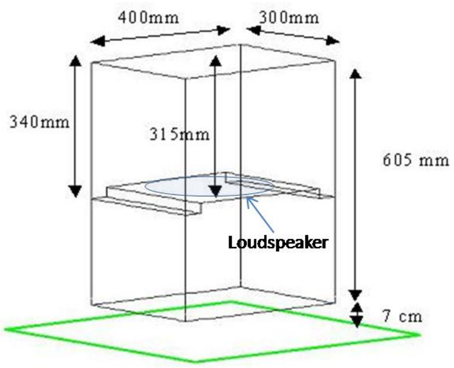


Figure 2. Schematic view of the box

The 50mm thick walls are made of double wood panels filled with sand. The test set-up is placed in a semi-anechoic room, whose walls are designed to absorb sound waves in the frequency range above 300Hz. Standing waves in this room may occur below this threshold.
The radiated power is calculated from intensity measurements, recorded using an intensity-probe. The box can be enclosed on the top using the sandwich panel. When this is mounted on the box, the panel has simply supported edges (clamping could damage the core material). Since the panel is not perfectly flat, plasticine is used to seal the edges. Therefore a certain unknown amount of structural damping is added to the system and the boundary conditions are not unambiguously known.

The surface above the sandwich panel is divided into 12 subsurfaces of 10cm x 10cm and then scanned with an intensity probe. The spacing between its two $\frac{1}{4}$ inch microphones is 5cm, making the measurements reliable up to 1250Hz. The sample frequency of the signals from these two microphones at the intensity probe is 5120Hz. The maximum measured frequency is 2560Hz. The frequency resolution is 1,25Hz, that corresponds to an observation time of 0,8s. Each measurement consists of 15 averages and is carried out three times (45 averages per subsurface). A Hanning window on all the signals is used to reduce the leakage. The experimental set-up is shown in Figure 3.

2.2 Experimental IL

The radiating surface is discretized into twelve subsurfaces. The sound power is calculated as:

$$\mathbf{P} = \sum_{i=1}^{12} I_i S_i \tag{2}$$

In this formula I_i is the measured intensity at subsurface "i" and S_i the size of the same subsurface. The intensity is measured with and without the enclosing sandwich panel set on the top

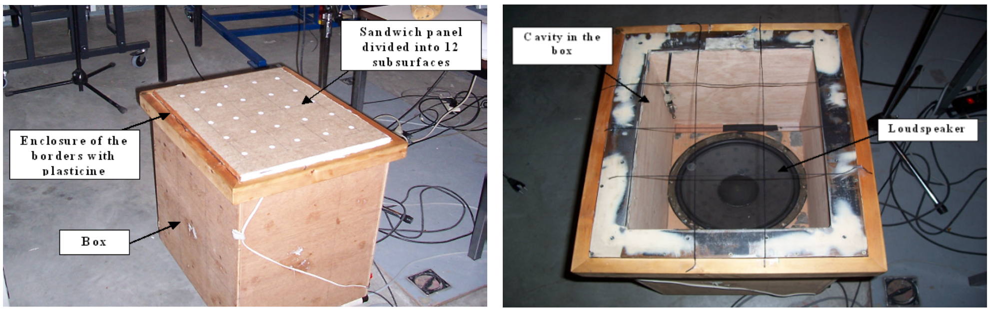


Figure 3. Test set-up

of the cavity. Figure 4 shows the power from both measurements in dB, i.e. the Sound Power Levels, L_P (reference power level of 10^{-12}W), and the resulting insertion loss, up to 700Hz.

$$\mathbf{L_P} = 10 \log \frac{P}{P_{ref}} \tag{3}$$

In the results shown in Figure 4 the modes of the coupled system (sandwich panel set on the

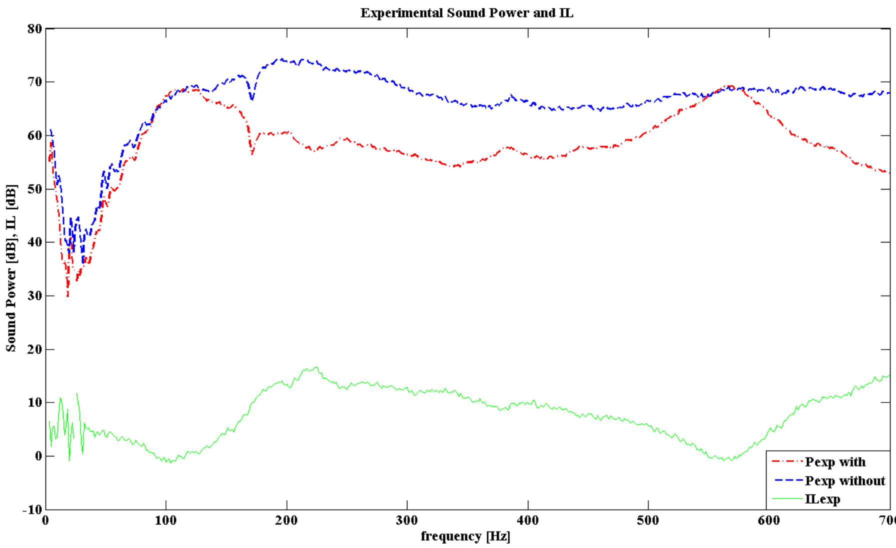


Figure 4. Experimentally determined sound powers and insertion loss [dBre10^{-12}W]

cavity) are crucial in determining the dip and peak locations in the resulting insertion loss. These modes can be either structurally or acoustically dominated, depending on the characteristics of the two subsystems. The first drop in the IL is located at around 100Hz and is caused by the first structural resonance of the sandwich panel. The coupled resonance frequencies must be known to understand better the location of the other drops and peaks in the insertion loss. A numerical model can be helpful for this purpose.

3. NUMERICAL STUDY OF INSERTION LOSS

The numerical model allows the analysis of the coupled modes and can also be useful for further investigations. Once optimized, it will allow a consistent reduction of the number of the experimental tests for the parametric determination of the IL (i.e., varying the geometrical parameters of the examined structure). The numerical model is presented in Figures 5 and 6.

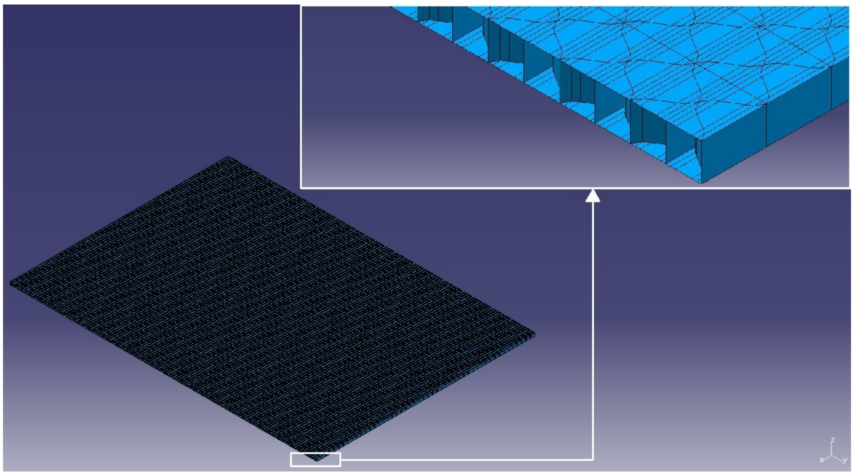


Figure 5. The detailed FE model of the sandwich panel

It consists of a structural finite element model of the sandwich panel (shown in Figure 5) coupled to an acoustic, indirect boundary element model of the acoustic cavity (proposed in Figure 6, with the applied boundary conditions).

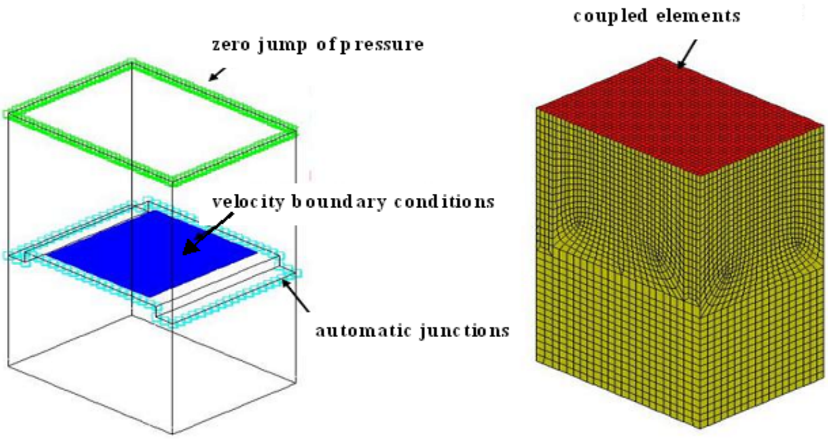


Figure 6: On the left side the BCs applied to the BE model – On the right side the BE model of the coupled system, cavity and enclosing panel

A lot of studies [10], have been already carried out to numerically study the coupled vibro-acoustic behaviour of structural-acoustic systems.
In this work LMS.VirtualLab Rev8b is used to calculate the radiated acoustic powers and MD NastranR3b for the prediction of the uncoupled structural dynamics.

3.1 The structural FE model

The structural part of the model, shown in Figure 5, consists of a detailed finite element model of the sandwich panel. The model is a sequence of unity cells that approximate the geometry of the core cells covered with face elements, given in details in Figure 7. The model of the entire panel, with dimensions of 496mm x 387mm, consists of 198810 bilinear 4-noded shell elements and 129214 nodes. It has been validated through an experimental modal analysis, carried out up to 300Hz on a freely suspended panel. Good correlation is found between the experimental and predicted natural behaviours, as shown in Table 2, with MAC (Modal Assurance Criterion)

Mode pair	Sim.	Freq. [Hz]	Exp.	Freq. [Hz]	Difference [%]	Exp. damping [%]
1	1	48,6	1	49,2	1,22	1,93
2	2	57,4	2	59,3	3,20	1,54
3	3	104,6	5	108,5	3,59	1,95
4	5	139,8	7	135,3	-3,32	2,59
5	6	162,8	8	164,2	0,85	1,89
6	7	215,4	11	211,4	-1,89	1,89
7	9	286,1	14	278,9	-2,58	2,28

Table 2: Comparison between numerical and experimental modal analysis results up to 300Hz, for a freely suspended panel. The damping coefficients come from the experimental modal test

Material	$E_{MD}[GPa]$	$E_{CD}[GPa]$	$\rho[kg/m^3]$	$\nu[-]$	thickness [mm]
skin	2	2	451	0,2	0,55
reinforced walls (core)	5	2	636	0,2	0,25
cellular walls (core)	4,76	1,6	636	0,2	0,1

Table 3. Material properties of the sandwich panel.

values larger than 70%.
An estimate of the structural damping comes from the experimental modal analysis of the sandwich panel examined, carried out up to 300Hz. The free modes result in a structural damping of about 2%. This value is used to model the structural damping, considered to be constant all over the analyzed frequency region [0-1,4]kHz.

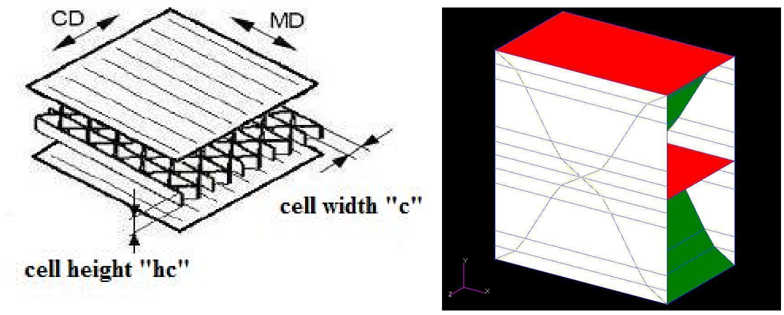


Figure 7: On the left side a schematic view of the panel structure – On the right side the unity cell model

The material properties come from previous works [11, 12], and are listed in Table 3. The real boundary conditions (test panel supported at the edges on the box and sealed with plasticine), are approximated in this numerical model by fixing the translational degrees of freedom of all the nodes at the edges of the lower skin.

3.2 The acoustical BE model

The acoustic part of the numerical model is an indirect, boundary element model of the box. The numerical simulations have to calculate radiated sound power both with and without the sandwich panel mounted on top of the box. This requires two models: one boundary element model of the closed box and one boundary element model of the open box. For the model to be accurate up to 2kHz, at least 10 linear elements per wavelength are required. This results in a

model with a total of 7288 elements. The structural modes of the supported detailed sandwich model are projected on a coarser mesh in order to get a model size requiring a reasonable, optimized computational effort. From the numerical modal analysis carried out up to 2kHz, the maximum number of bending wavelengths is 4 in the length (at 1991,4Hz) and 3 in the width (at 1995,5Hz). The use of at least 10 linear elements per bending wavelength for this model in the range up to 2kHz is required for a good accuracy (a rule of thumb).
Thus, the simpler mesh, on which the structural results are projected, has 40 elements in the length and 30 elements in the width (1200 elements in total). The 1200 elements on top of the cavity are absent for the simulation of the sound power without the sandwich panel. They are coupled to the elements of the simple structural mesh with the projected modes for the simulations of the sound power with the panel.
There are three sets of boundary conditions as shown in Figure 6. The first set contains the automatic junctions. This condition takes care of a split up of the nodes at T-crossings to ensure normal vector consistency. The second set contains the velocity boundary conditions that simulate the vibrating membrane of the loudspeaker. The velocity is one directional into the cavity and its magnitude is 1m/s. The third set is a zero jump of pressure at the top edge nodes of the box cavity for the determination of the sound power of the opened box.
The real test set-up is placed in a semi-anechoic room, with a rigid floor. Because of this a plane of symmetry must be added in the model at 7cm below the box. The whole semi-anechoic room (and thus, the effects of the standing waves generate and propagate in it below 300Hz) is not included into the numerical model.
In order to determine the sound power from the calculated potentials, a set of field points has to be defined. This consists of a plane 5cm above the sandwich panel. It does not contain all the radiated sound power, because part of it can escape from the edges of the plane (not included in it), but it is a better approximation of the real set-up, because the sound power calculated from the intensities is evaluated at the panel surface.

3.3 First numerical results

The structural and acoustic modes are necessary in order to understand the coupled behaviour. The structural modes are calculated from the detailed, simply supported structural finite element model. The first ten modes are shown in Table 4. In order to obtain a satisfying accuracy up to 1,4kHz, modes up to 2kHz are calculated. To determine the acoustic modes, an acoustic finite element model of the cavity is built. The first six modes of the closed cavity are shown in Table 5 (in the same frequency range). When the coupling of the structural model and the acoustic model is taken into account (here a fully-coupled system has been considered), new coupled modes and resonance frequencies exist, which are structurally or acoustically dominated. Table 6 sums the frequencies of the first twelve coupled modes and the corresponding dominant mode.

It is expected that the modes dominated by the first, the fourth and the seventh structural mode (enclosed cavity case) will increase the radiation of sound power. Their frequencies are respectively 127Hz, 285Hz and 452Hz (see Table 6). This depends on their structural mode shape. The other structural modes always have an even number of bending wavelengths in the length and/or in the width direction. So, for each air particle that is moved by the panel, there is a neighbouring particle that has an opposite movement (as seen in Table 4). In this way sound power is not radiated. For the first, fourth and seventh structural mode there is an odd number of bending wavelengths. Another mode that strongly improve the radiation of sound power is that one dominated by the second acoustic mode. This is the mode of the vertical standing wave in the acoustic cavity. It will excite the panel in the strongest way. Its coupled frequency is 553Hz.

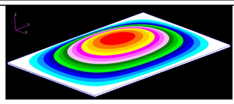
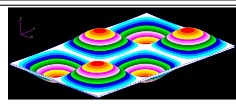
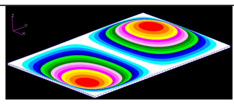
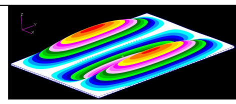
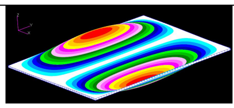
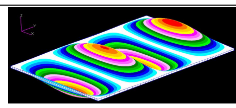
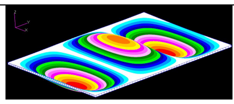
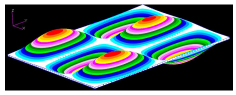
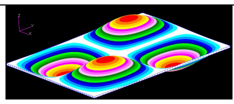
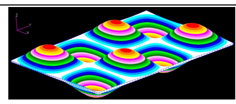
Mode	Frequency [Hz]	Mode shape	Mode	Frequency [Hz]	Mode shape
1	91,4		6	435,4	
2	163,8		7	461,8	
3	218,6		8	464,7	
4	292,3		9	538,3	
5	305,6		10	619,95	

Table 4. First ten uncoupled structural modes of the supported sandwich panel.

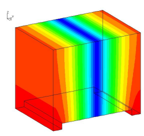
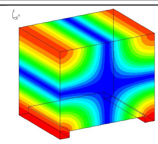
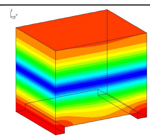
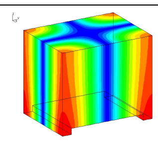
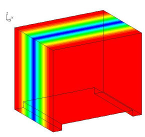
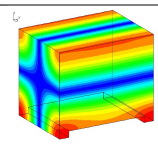
Mode	Frequency[Hz]	Mode shape	Mode	Frequency [Hz]	Mode shape
1	418,2		4	664,9	
2	530,8		5	704,5	
3	566,9		6	776,6	

Table 5. First six uncoupled acoustic modes of the closed cavity.

Mode	Frequency [Hz]	Dominant mode	Mode	Frequency [Hz]	Dominant mode
1	126,9	1st struct (91,4Hz)	7	426,5	1st acoustic (418,1Hz)
2	154,1	2nd struct (163,8Hz)	8	452,3	7th struct (461,9Hz)
3	209,2	3th struct (218,6Hz)	9	459,8	8th struct (464,7Hz)
4	285,3	4th struct (292,3Hz)	10	529,8	9th struct (538,4Hz)
5	295,8	5th struct (305,6Hz)	11	552,9	2nd acoustic (530,8Hz)
6	425,8	6th struct (435,4Hz)	12	576,3	3th acoustic (566,9Hz)

Table 6. First twelve coupled modes. In brackets the reference uncoupled frequency.

The first peak in the radiated sound power without the panel will be at the first vertical acoustic mode of the open cavity. Figure 8 shows the numerical sound power without the panel, the sound power with the panel and the resulting insertion loss up to a frequency of 700Hz. The first vertical mode of the opened box is situated at approximately 200Hz. The four peaks in the radiated power for the cavity enclosed by the panel are at 127Hz, 285Hz, 452Hz and 553Hz and result in dips of the insertion loss at these frequencies (as explained above).

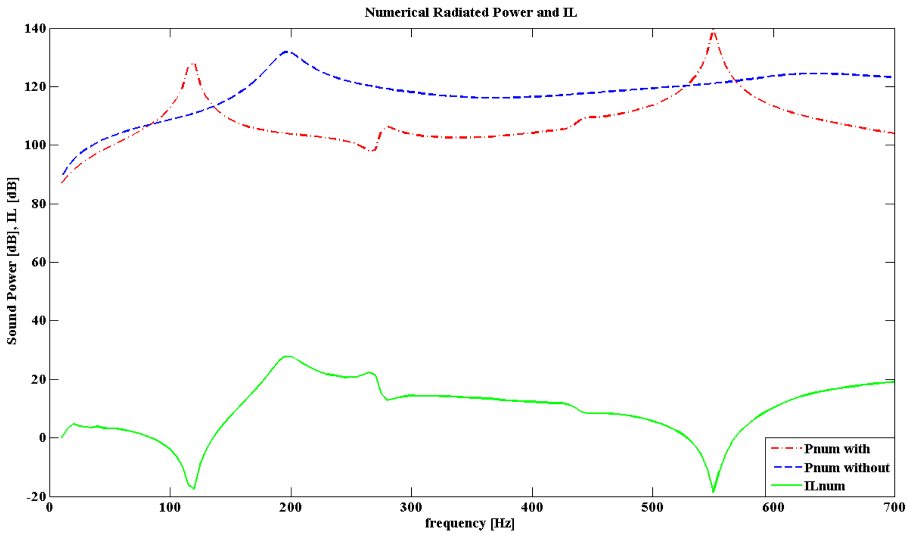


Figure 8. Numerically determined sound powers and insertion loss [dBre 10^{-12}W]

4. IL COMPARISON

In Figures 9 and 10 the experimental and numerical sound powers and the insertion losses are respectively, compared up to 1,4kHz. Although the numerical sound powers are higher than the experimental ones (due to a bigger simulated excitation used in the numerical model), the absolute values of these powers are not influencing on the insertion loss calculation (since the latter comes from their difference).

The peaks' locations in the numerical powers correspond to those of the experimental powers. At 550Hz there is a drop in the insertion loss, caused by the peak in the power with panel. The location of this drop is a bit shifted (20Hz in advance in the numerical simulation). This is due to the finite element model updated for modes only up to 300Hz, so less accurate at the higher frequencies.

It is clear that the peaks in the experimental sound powers are wider then those from the numerical simulations. This indicates the numerical structural damping used (2%, coming from the experimental modal analysis on the panel suspended in free-free conditions) is too low. Actually, in this model any additional damping introduced into the system by the use of the plasticine (to seal the edges when the panel is set on the cavity) is discarded.

The experimental sound power levels with and without the panel are higher than the numerical simulations at about 150Hz. The peaks around this frequency appear quite wide. The first structural mode (127Hz) and the first acoustic one (open cavity, 200Hz) are supported by an additional mode, found only experimentally and in both the open and the enclosed cavity cases. It occurs at 150Hz and is related to the standing waves in the semi-anechoic room. The absorption of this room is not effective below 300Hz. Its effect results clear in the experimental measurements (Figure 9), where it makes the peaks at 127Hz and 200Hz wider. The room is not included in the numerical model, so the numerical results in the same frequency band only show the two well separated peaks (127Hz and 200Hz).

Apart from the huge amplifications occurring in the predicted IL at the occurences of those strongly radiating modes (i.e., in the coupled system, 127Hz, 285Hz , 452Hz, etc..), a general good agreement between this and the experimental results is found all over the frequency range, except in some regions, like around 900Hz. Looking at the radiated power of the open box in Figure 9, this can be intuitively addressed to the not yet update acoustic damping. Considering an effective absorption of the box walls (finite acoustic impedance), the real behaviour in these bands could be better predicted.

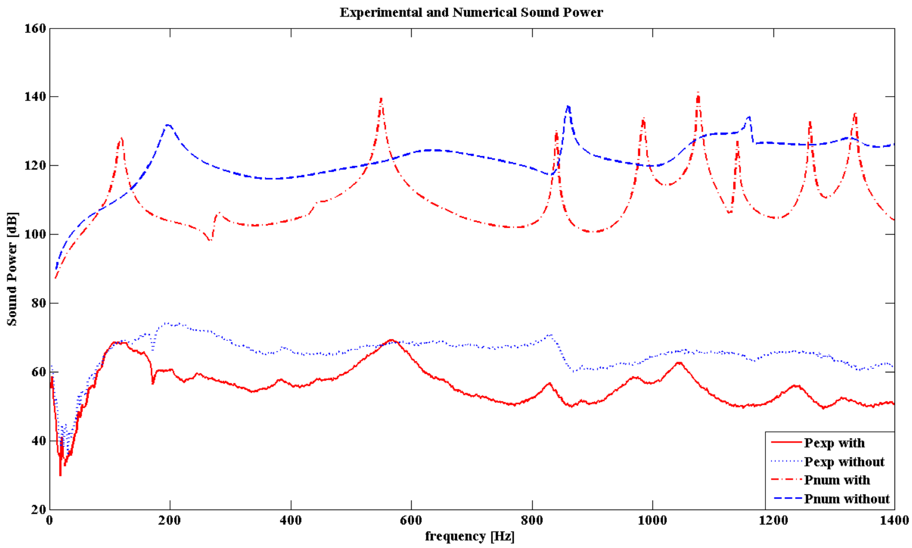


Figure 9. Radiated Power comparison

5. CONCLUSIONS

This paper studies the insertion loss of a sandwich panel with a TorHex paper honeycomb core and faces of PP/natural fibres up to 1,4kHz. The insertion loss, determined experimentally and numerically, has a mean value of 12dB up to 1,4kHz.

The experimental analysis is performed using the developed test set-up, consisting in a box containing a loudspeaker and an acoustic cavity, located in a semi-anechoic room. The insertion loss is influenced by the acoustic modes of the attached cavity, as is also the case in most

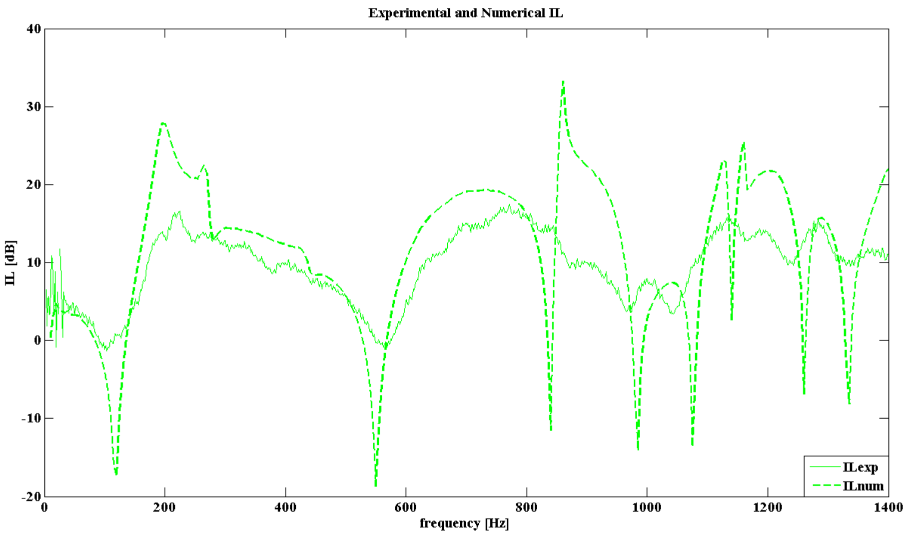


Figure 10. Numerical and Experimental IL comparison

real-life applications. The aim of the numerical model of the cavity enclosed by the sandwich model is to develop a useful tool, to facilitate the determination of the insertion loss after adaptations of the panel design. The model consists of a structural finite element model of the panel coupled to an acoustic, indirect boundary element model of the cavity. The numerical results are compared to the experimental results. Some obvious differences are highlighted. Looking at the experimental results, the contributions to the acoustic radiation, coming from the first structural mode of the sandwich panel (127Hz) and from the first acoustic mode of the open cavity (200Hz), seem to be quite uniformly spread all over the frequency band between these two natural frequencies. This is due to a natural behaviour of the semi-anechoic room, not included in the numerical simulation and whose absorption is not effective below 300Hz. For frequencies higher than 300Hz there are shifts in the location of dips and peaks in the IL, caused by the limited updating of the structural model. All over the considered frequency region the trend is well caught, although the amplitude level is quite often overestimated. Updating the damping coefficient and simulating more accurately the acoustic properties of the test set-up, a new optimized model will be developed and it will be a useful tool for prediction of the insulation properties of such a complex structure.

ACKNOWLEDGMENTS

The authors kindly acknowledge the European Commission for their support of the Marie Curie EST project SIMVIA2 (http://www.simvia2.eu, contract nr. MEST-CT-2005-020263), from which Miss. Marianna Vivolo holds a Research Training grant. Also the Fund for Scientific Research - Flanders (F.W.O.), Belgium, is gratefully acknowledged for their research support.

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